RATIONAL DESIGN OF STEAM HEADERS AND PIPING SYSTEMS

BY

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B. S. University of Illinois, 1917

THESIS

Submitted in Partial Fulfillment of the Requirements for the

Degree of

MECHANICAL ENGINEER

IN

THE GRADUATE SCHOOL

OF THE

UNIVERSITY OF ILLINOIS

1922

UNIVERSITY OF ILLINOIS

THE GRADUATE SCHOOL

April 18, 1922

I HEREBY RECOMMEND THAT THE THESIS PREPARED BY

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ENTITLED "Rational Design of Steam Headers and Piping Systems".

BE ACCEPTED AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE

PROFESSIONAL DEGREE OF Mechanical Engineer

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RATIONAL DESIGN OF STEAM HEADERS AND PIPING SYSTEMS.

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RATIONAL DESIGN OF STEAM HEADERS AND PIPING SYSTEMS

1 - SCOPE AND FOREWORD.

The scope of this thesis is limited to that part of design which deals with (a) the selection of heat insulating coverings, and (b) the proper pipe sizes. The remaining two points to be considered in the design of a steam header, namely, the location of the pipe, and provision for expansion, are thoroughly treated in text books and will not be considered herein.

It would seem at the outset that the logical order of treatment would be to first lay out the piping scheme and determine the proper sizes. Having done so, the last item would be to determine the thickness of insulation to use on the pipe. Under the "rational" method of design, however, the reverse order becomes better suited to the solution. The determination of pipe size involves the amount of heat radiated from the system, which in turn, involves the thickness of insulation. Further, it is good practice to set up an "insulation schedule" which indicates the economical thickness of insulation for all sizes of pipe which may occur in the whole plant. By drawing up this schedule first, the proper thicknesses of insulation will be known for the several pipe sizes considered in the calculations for the economical pipe size.

2 - REVIEW AND CRITICISM OF METHODS USUALLY FOLLOWED IN THE SELECTION OF INSULATION.

Conversation upon the subject has indicated the surprising prevalence of the idea among designers that "a little insulation does some good, and the more the better", or, as one designer stated, "Modern plants use extreme temperatures and the radiation losses become enormous; consequently, too much insulation cannot be used".

The fallacy of this idea lies in the fact that the heat saved is not directly proportional to the thickness of the insulation. One inch of insulation under average conditions saves about 89 per cent. of the heat, which would be lost from the bare pipe. Three inches saves about 94 per cent. of the bare pipe loss; so it is seen that the last two inches of insulation, which costs about two times as much as the first inch, is only 5.6 per cent. as effective in saving heat.

The Magnesia Association of America publishes four charts, based upon four values of steam cost, each chart indicating the proper thickness of 85% magnesia to use for each pipe size and for various temperature differences. The steam costs represented by the four charts are:- 20¢, 40¢, 60¢, and 80¢ per million Btu. When steam costs do not coincide with one of these four values, the result must be obtained by observing the two charts for lesser and greater steam costs, and approximating the intermediate value. This method does not permit of a satisfactorily definite solution. A more important criticism of these charts, is the fact that the curves are regular curves without inflections. In the rational method of analysis it will be found that these curves are not regular, due to the fact that the list prices for various thicknesses do not follow a general curve. Hence the accuracy of such charts is to be doubted.

Johns-Manville, in their bulletin "Service to Power Users", publish a brief table which indicates the minimum thickness of steam pipe insulation that should be used for a given character of service, e. g. 25 to 100 pounds steam pressure, 100 to 200 pounds, etc. The results obtained from interpolation in this table are questionable, and the "minimum" thickness does not imply "economical" thickness at all.

3 - PROPOSED RATIONAL METHOD FOR SELECTING INSULATION.

By "Rational method" is meant that method which effects the highest economic efficiency, and "highest economic efficiency" means the lowest total financial charges against the system under consideration. The rational method, then, consists of an analysis of the costs against the system, and the determination of the condition which corresponds to the minimum value of the total costs.

4 - COST FACTORS INVOLVED IN THE SELECTION OF INSULATION.

The costs which are chargeable against the installation are:

a - Fixed charges -

b - Interest on cost of insulation, in place -
c - Effect of nature of material.
d - Effect of thickness of material.
e - Depreciation of covering.
f - Repairs and maintenance.
g - Miscellaneous (taxes, insurance, etc.)

h - Operating charges -

i - Cost of heat radiated -
j - Effect of nature of material.
k - Effect of thickness of material.

NOTES:
Item
a Fixed charges include those which continue throughout the useful life of the material, whether the plant operates or not.

c Some materials, having higher insulating efficiency than others, naturally command higher prices, and the interest on the installation must correspond to the first cost.

d Thick coverings naturally are more costly than thin coverings, and again higher insulation efficiency is accompanied by higher interest charges.

h Operating charges are those which occur only when the plant is in operation. Most plant operating charges (e. g. fuel, water, lubricants) vary in direct proportion with the quantity of product produced. Radiation from insulated pipe, however, is practically a constant amount, regardless of the rate of plant operation.

i, j and k. The amount of heat radiated is proportional to the complement of the insulating efficiency of the covering. The efficiency is higher

for some materials than for others, and increases somewhat with thickness.

5 - GRAPHIC METHOD OF COMPARING INSULATION CHARGES.

The manipulation of data for the preceding outline is best effected by a tabular arrangement, but the results, costs, are most easily compared and analysed by a graphic presentation. Since thickness is the argument, it is plotted as abscissa. Ordinates are costs, and three cost curves may be plotted for each pipe size:

(1) Annual fixed charges on insulation,
(2) Annual cost of heat radiated,
(3) Total annual cost.

The minimum value of the total cost curve is the criterion for the economical thickness of covering. If peculiar inflections occur in the total cost curve, the source may be discovered by a glance at the two component curves.

6 - METHODS RECOMMENDED BY DESIGNERS AND TEXT BOOK WRITERS FOR PROPER PIPE SIZE DETERMINATION

Two general methods of selecting pipe sizes have been in common use, namely, the velocity method, and the pressure loss method.

The velocity method consists in selecting pipe of such size that the velocity of the steam will not exceed certain values. Undoubtedly this rule is based upon the theory that velocities exceeding the prescribed values would result in excessive pressure loss and consequent poor economy.

This theory is only partially borne out in practice. Pressure loss varies as the square of the velocity, when the weight of steam flowing is the variable, and it is found that for velocities much in excess of 10,000 feet per minute that the pressure loss per 100 feet of pipe becomes so great that it is excessive for long runs of pipe. But the average steam header layout is of such design that numerous branch connections, at which steam is fed into or bled out of the header, effect frequent changes of velocity, such that one or more sections of the header may contain steam at almost zero velocity, while another section may carry steam at an extremely high velocity. It would not be proper to increase the size of the section which carries high velocity steam, because it is possible that a shifting of the load, by replacing one or more of the operating boilers by others which have not been in operation, may almost reverse the values of velocity in the two critical sections mentioned. For this reason, it is advisable in most cases to construct the entire header of one size of pipe, and it is seen that if the size were so selected that the maximum velocity in any section is less than 10,000 feet per minute, then the velocities in all the other sections would be too low, due to oversize pipe, resulting in excessive superheat loss, or condensation, and capital charges.

Further, the velocity which corresponds to the maximum efficiency in one case may be very different from that in another case. For instance, the relative location of boilers and prime movers, as "back to back", "end to end",

etc, has a decided effect upon the characteristics of the system, and upon the velocity which corresponds to maximum economy.

The pressure loss method consists in selecting pipe of such size that the pressure loss in the system will not exceed certain values. The principal difficulty in applying this method is in knowing what pressure loss corresponds to maximum economy. Different designers recommend from five to 20 pounds, and occasionally the theory is advanced that a loss of 50 pounds or more is not objectionable, since the energy remains in the steam as additional superheat. The latter recommendation of course is based upon a misinterpretation of fact, because although the expansion is practically a constant-total-heat process, the entropy increases during the expansion and the percentage of availability of the energy decreases, so that less is available for conversion in a prime mover, and more must be lost in the exhaust.

It is impossible to assign a definite value for the velocity or pressure loss which will correspond to maximum economy in all cases, since there are so many factors, such as type of header layout, type of prime mover, station load factor, etc which affect the problem.

7 - PROPOSED RATIONAL METHOD FOR SELECTING PIPE SIZES.

"Rational Method", as previously explained, indicates that method which effects the lowest total financial charges against the system under consideration. The rational method of determining economical pipe size, then, consists in analysing the costs against the system with various sizes of pipe, and finding the size which corresponds to the minimum value of total costs.

8 - COST FACTORS INVOLVED IN THE SELECTION OF PIPE SIZE.

The costs which are chargeable against the installation are:

Fixed charges -

Interest on cost of pipe, fittings, and insulation, erected.
Depreciation of pipe and insulation.
Repairs and maintenance.
Taxes and insurance.

Operating charges -

Annual cost of steam to operate prime mover.
Annual cost of heat radiated.

9 - GRAPHIC METHOD OF COMPARING PIPE SIZE DATA.

The compilation of data for the preceding outline is best effected by a tabular arrangement, but the variation of the several factors which affect the costs, as well as the costs themselves, are most easily compared and analyzed by a graphic presentation. Since pipe size is the argument, it is plotted as abscissa. Ordinates represent costs, velocities, etc, to suit the factors plotted. The following items may be shown to advantage:

Steam velocity
Pressure loss due to friction

Pressure loss due to velocity head.
Total pressure loss.
Water rate of prime mover.
Annual cost of steam for prime mover.
Annual fixed charges on pipe, fittings, and
insulation.
Annual cost of heat radiated.
Total annual costs.

APPENDIX I

Design of a Simple Steam Header.

The plant chosen for this example is Boiler House "L" at National Works of National Tube Company at McKeesport, Pennsylvania. This installation was completed and placed in operation in May 1919.

10 - DESCRIPTION. (See Fig. 1)

The plant consists of one - 10,000 kw. (80% power factor) Curtis turbo-generator served by two - 1471 horse power Babcock and Wilcox cross-drum boilers. The plant is located in a space between two rolling mills, and since there is no room for future extension, additional capacity need not be provided for in the design of piping, etc. Also, since the steam conditions differ from those employed in the general mill system, the plant may be treated as an isolated unit.

11 - STEAM CONDITIONS.

Normal gage pressure at boiler drum - 250 lb. per sq. in.
Normal superheat - 150° F.
Temperature of saturated steam - 406° F.
Temperature of superheated steam - 556° F.
Temperature of air (assumed) - 80° F.

12 - LOAD CONDITIONS.

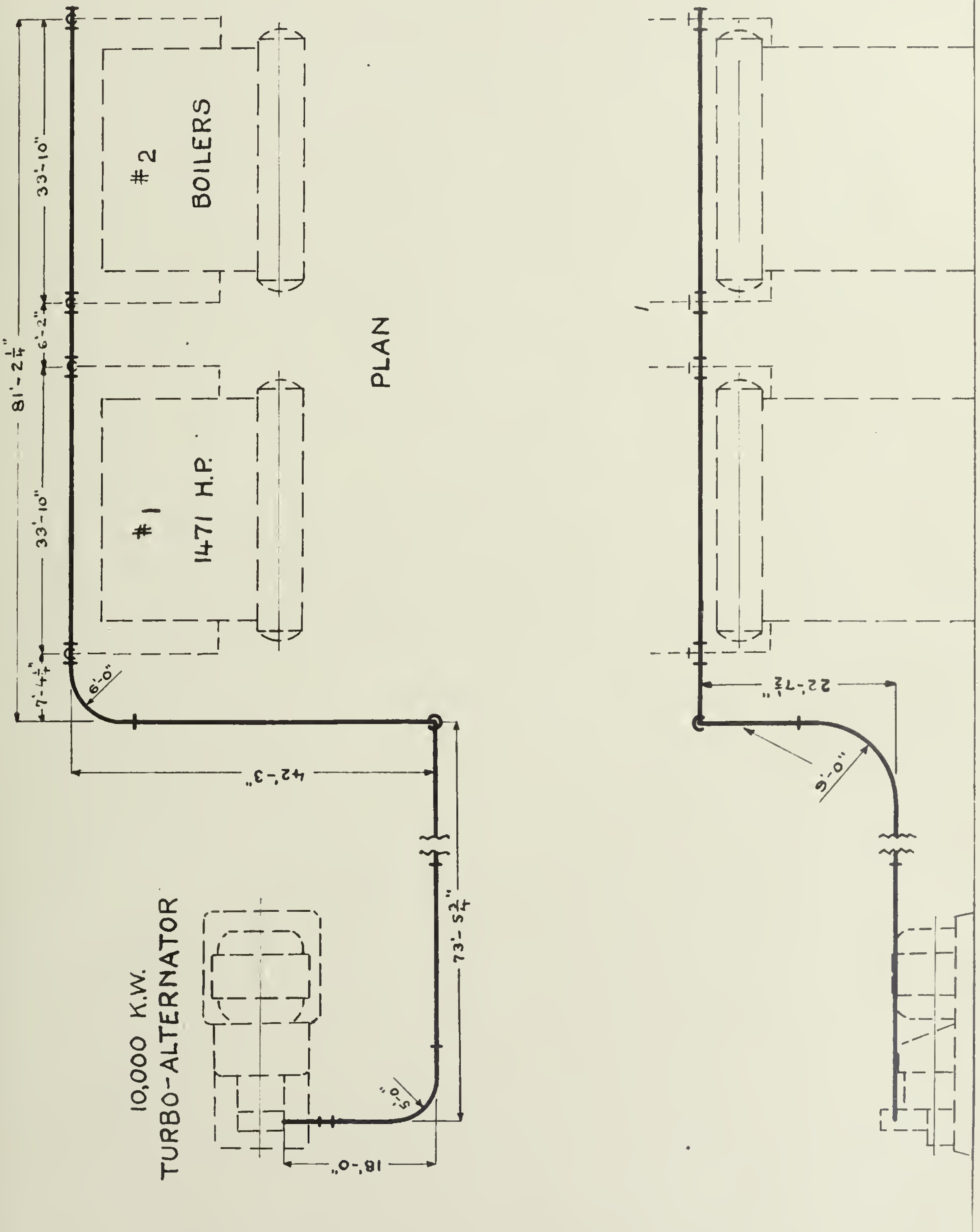
The turbo-generator is intended primarily to serve seven motor-generator and rotary converter sub-stations, distributed about the mill. It is also connected, through transformers and a high-tension transmission system, to other power generating and consuming equipment at plants of other subsidiary companies of the United States Steel Corporation. In designing the plant, it was anticipated that the normal load on the generator would be 8000 kw., and that this load would be carried during 6400 hours per year. During the remaining 2360 hours, when the rolling mills etc, are not operating, power may be received over the high tension system, from generating stations operating with blast furnace gas or waste heat as fuel.

13 - STEAM REQUIREMENTS.

The water rate of the turbine at 8000 kw. output is 12.9 lb. per kwh. with the steam conditions enumerated.

With the steam required for auxiliaries taken from the end of the header farthest removed from the turbine, the only load which need be considered in the design of the header proper is that required for the turbo-generator.

At the design load, and under design conditions, the steam required for the turbine = 8000 x 12.9 = 103,200 lbs. per hr.



— FIG. 1. —

— LAYOUT OF A SIMPLE STEAM HEADER. —

14 - SELECTION OF INSULATION.

Previous studies have indicated that the total annual charges on 85% Magnesia covering are about four percent lower than on Asbesto-Sponge-Felted; hence, from the standpoint of cost alone, 85% Magnesia would be recommended. However, Asbesto-Sponge-Felted is considered more durable, more easily removed and replaced, and less liable to deterioration from de-hydration at steam pipe temperatures. These advantages cannot easily be capitalized, but they are considered to more than outweigh the four percent difference in annual costs, and Asbesto-Sponge-Felted covering will be used in this analysis.

15 - ANALYSIS OF COSTS.

Table No. 1 is the tabular solution of economical thickness for the insulation. The table, together with the explanatory notes which follow it, is self-explanatory.

Fig. 2 is a graphic presentation of items 4, 6 and 7 in the table and shows the trend of the component cost lines, as well as the deciding factor, total cost.

16 - EXPLANATION OF ITEMS IN TABLE NO. 1.

Item 1.

By interpolation and extrapolation from Table 5, Appendix III.

Item 2.

With 13,500 Btu. coal at $3.50 per net ton and 78% boiler efficiency, the cost of heat per million Btu. at the boiler nozzle =

$$\frac{1{,}000{,}000 \times 3.50}{13{,}500 \times 0.78 \times 2000} = \$0.166 \text{ per million Btu.}$$

$$\text{Item 2} = \frac{\text{Item 1} \times 8760 \times 0.166}{1{,}000{,}000} = 0.001455 \times \text{Item 1.}$$

Item 3.

By interpolation and extrapolation from Table 6, Appendix III.

Item 4.

Item 4 = Item 1 x (100 - item 3) ÷ 100.

Item 5.

List prices from Table 7, Appendix III.
Net prices = list less 30% (quoted January 19, 1922 by H. W. Johns-Manville Co.)

Item 6.

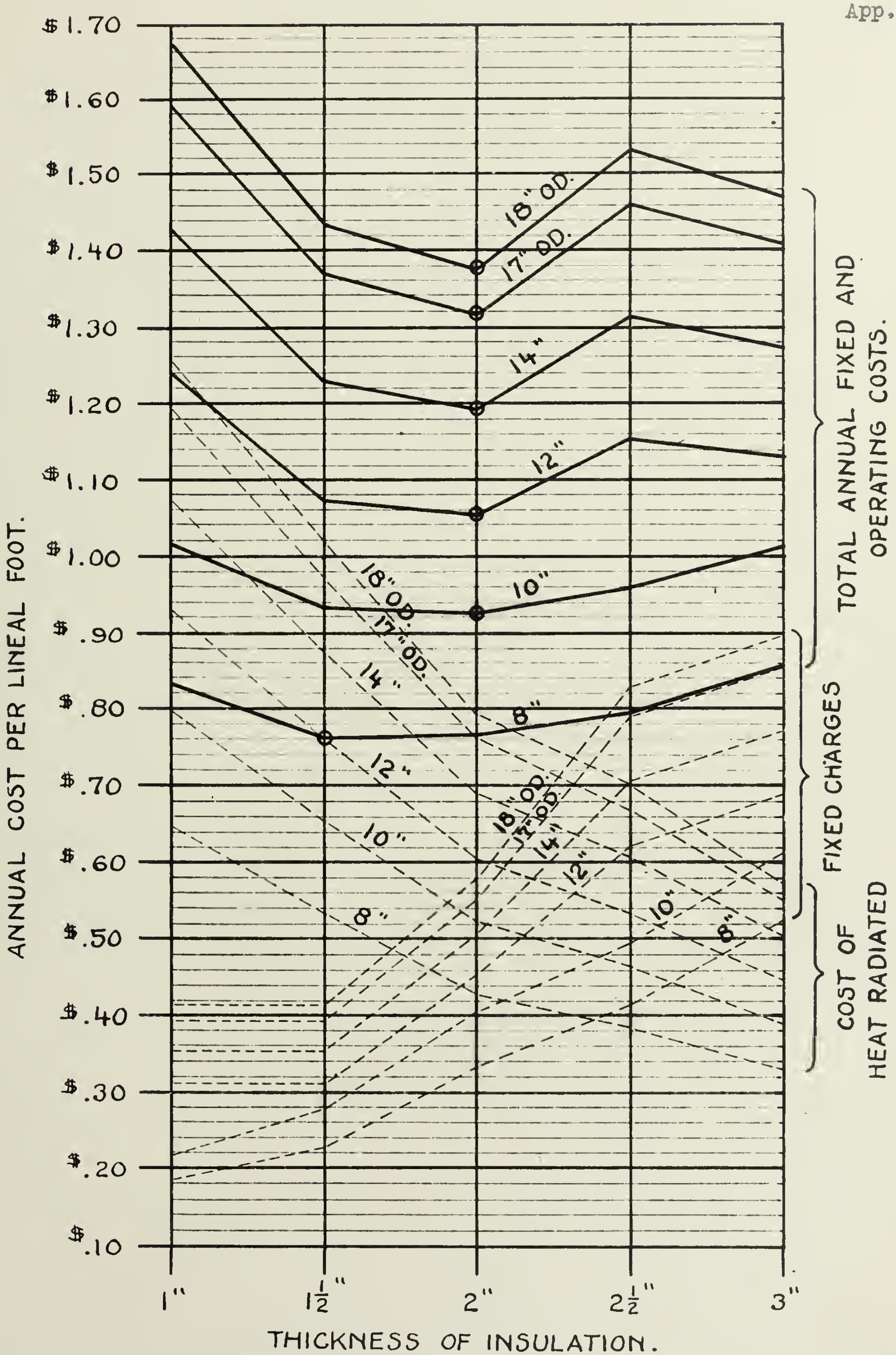
Item 6 = (item 5 + 60% for labor and accessories) x 0.15 =

Table No. 1
Selection of Steam Pipe Insulation.
Asbesto-Sponge-Felted.
475° Temperature Difference.
All Quantities are per Lineal Foot.

Item Number	Thickness of Insulation	1-Inch	$1\frac{1}{2}$-Inch	2-Inch.	$2\frac{1}{2}$-Inch	3-Inch
	Column Number	1	2	3	4	5
		8-Inch Pipe				
1	Bare Pipe loss - Btu. per hr. per lineal foot	5239	5239	5239	5239	5239
2	Cost of heat lost per year - bare pipe	$7.52	$7.52	$7.52	$7.52	$7.52
3	Efficiency of Asbesto-Sponge-Felted - %	91.40	92.90	94.30	94.90	95.70
4	Cost of heat lost per year - covered pipe	$0.646	$0.534	$0.428	$0.383	$0.323
5	Cost of covering only - net	$0.77	$0.94	$1.40	$1.72	$2.20
6	Annual fixed charges	$0.185	$0.226	$0.336	$0.412	$0.528
7	Total annual fixed and operating charges.	$0.831	$0.760	$0.764	$0.795	$0.851
		10-Inch Pipe				
1	Bare Pipe loss - Btu. per hr. per lineal foot	6538	6538	6538	6538	6538
2	Cost of heat lost per year - bare pipe	$9.50	$9.50	$9.50	$9.50	$9.50
3	Efficiency of Asbesto-Sponge-Felted - %	91.60	93.10	94.50	95.10	95.90
4	Cost of heat lost per year - covered pipe	$0.798	$0.656	$0.522	$0.465	$0.389
5	Cost of covering only - net	$0.91	$1.16	$1.68	$2.06	$2.55
6	Annual fixed charges	$0.218	$0.278	$0.403	$0.494	$0.612
7	Total annual fixed and operating charges	$1.016	$0.934	$0.925	$0.959	$1.001
		12-Inch Pipe				
1	Bare Pipe loss - Btu. per hr. per lineal foot.	7750	7750	7750	7750	7750
2	Cost of heat lost per year - bare pipe	$11.26	$11.26	$11.26	$11.26	$11.26
3	Efficiency of Asbesto-Sponge-Felted - %	91.76	93.26	94.66	95.26	96.06
4	Cost of heat lost per year - covered pipe	$0.927	$0.758	$0.601	$0.534	$0.444
5	Cost of covering only - net	$1.30	$1.30	$1.89	$2.59	$2.87
6	Annual fixed charges	$0.312	$0.312	$0.454	$0.622	$0.688
7	Total annual fixed and operating charges	$1.239	$1.070	$1.055	$1.156	$1.132

Table No. 1 - (Cont'd.)
Selection of Steam Pipe Insulation.
Asbesto-Sponge-Felted
475° Temperature Difference.
All Quantities are per Lineal Foot.

Item Number	Thickness of Insulation	1-Inch	1½-Inch	2-Inch	2½-Inch	3-Inch
	Column Number.	1	2	3	4	5
		14-Inch Pipe				
1	Bare Pipe loss - Btu. per hr. per lineal foot	9130	9130	9130	9130	9130
2	Cost of heat lost per year - bare pipe	$13.27	$13.27	$13.27	$13.27	$13.27
3	Efficiency of Asbesto-Sponge-Felted - %	91.92	93.42	94.82	95.42	96.22
4	Cost of heat lost per year - covered pipe	$1.072	$0.874	$0.687	$0.608	$0.502
5	Cost of covering only - net	$1.47	$1.47	$2.10	$2.94	$3.22
6	Annual fixed charges	$0.353	$0.353	$0.504	$0.705	$0.772
7	Total annual fixed and operating charges	$1.425	$1.227	$1.191	$1.313	$1.274
		17-Inch O.D. Pipe				
1	Bare Pipe Loss - Btu. per hr. per lineal foot	10,330	10,330	10,330	10,330	10,330
2	Cost of heat lost per year - bare pipe	$15.02	$15.02	$15.02	$15.02	$15.02
3	Efficiency of Asbesto-Sponge-Felted - %	92.04	93.54	94.94	95.54	96.34
4	Cost of heat lost per year - covered pipe	$1.196	$0.970	$0.760	$0.670	$0.550
5	Cost of covering only - net.	$1.64	$1.64	$2.31	$3.29	$3.57
6	Annual fixed charges	$0.394	$0.394	$0.554	$0.790	$0.857
7	Total annual fixed and operating charges	$1.590	$1.364	$1.314	$1.460	$1.407
		18-Inch O.D. Pipe				
1	Bare Pipe Loss - Btu. per hr. per lineal foot	10,940	10,940	10,940	10,940	10,940
2	Cost of heat lost per year - bare pipe	$15.90	$15.90	$15.90	$15.90	$15.90
3	Efficiency of Asbesto-Sponge-Felted - %	92.10	93.60	95.00	95.60	96.40
4	Cost of heat lost per year - covered pipe	$1.256	$1.018	$0.796	$0.700	$0.572
5	Cost of covering only - net	$1.73	$1.73	$2.42	$3.46	$3.74
6	Annual fixed charges	$0.415	$0.415	$0.580	$0.830	$0.898
7	Total annual fixed and operating charges	$1.671	$1.433	$1.376	$1.530	$1.470



—FIG. 2.—

—GRAPHIC STUDY OF INSULATION THICKNESS.—

= item 5 x 0.24.

The fixed charges are assumed to be : interest 6%, depreciation 5%, repairs and maintenance 1%, taxes, insurance, etc, 3%. Total 15%.

Item 7.

Item 7 = item 4 + item 6.

17 - CONCLUSIONS ON INSULATION.

The curves or item 7 of the table indicate that 2 inches is the economical thickness for all sizes from 10-inch to 18-inch pipe inclusive, and that 1-1/2 inches is the most economical for 8-inch pipe. However, as the difference between 1-1/2 inches and 2 inches on 8-inch pipe is only four mills per year per foot, it is considered preferable to simplify the specification by adopting 2 inches as the thickness for the six sizes of pipe considered.

18 - SELECTION OF HEADER SIZE.

Table No. 2 is the tabular solution of the header size determination. A complete set of explanatory notes follows the table.

Fig. 3 is a graphic presentation of the important items of the table. The curves are numbered to correspond with the items they represent.

19 - EXPLANATION OF ITEMS IN TABLE NO. 2.

Item 1.

From quotation submitted by Pittsburgh Valve Foundry and Construction Co. March, 1922.

Item 2.

From quotation submitted by H. W. Johns-Manville Company dated February 17, 1922.

Item 3.

Item 3 = 0.15 (item 1 + item 2). The fixed charges are assumed to be: interest 6%, depreciation 5%, repairs and maintenance 1%, taxes, insurance, etc, 3%. Total 15%.

Item 4.

Cost of heat lost = length of header (i. e. 202 ft.) x item 4, column 3 Table 1, Appendix I.

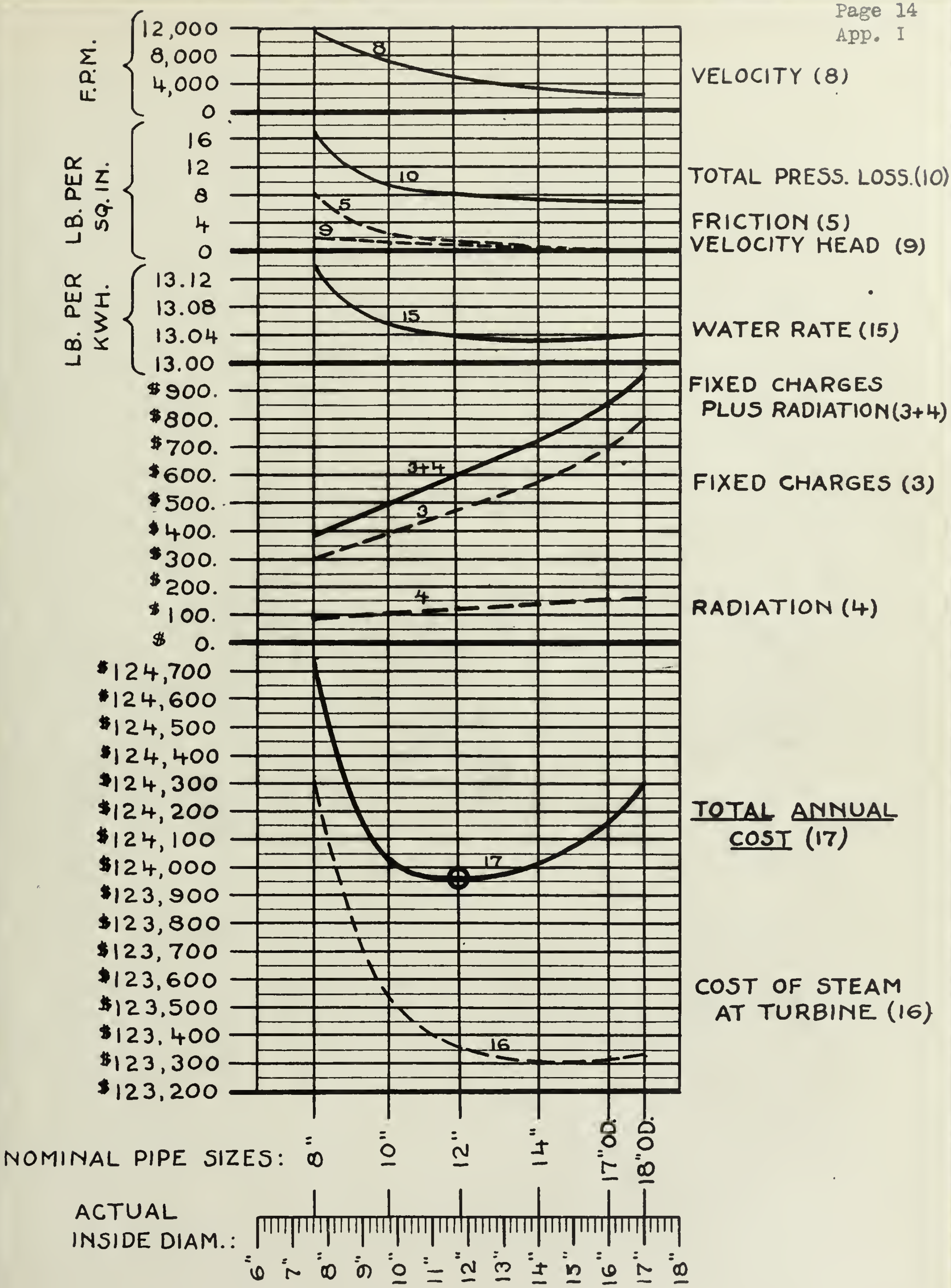
Item 5.

Use formula (2), Appendix IV.
$p = W^2 L V F$
$w = 103,200 \div 60 = 1720$ lb. per min.
$L = 165$ ft. (from point midway between boilers).

Table No. 2

Determination of Economical Header Size.

Item Number.	Nominal Size of Pipe, Inches.	8	10	12	14	17 O.D.	18 O.D.
	Column Number	1	2	3	4	5	6
1	Cost of header incl. fittings, erected	$1,375	$1,800	$2,340	$2,973	$3,566	$4,335
2	Cost of insulation, incl. fittings, placed	$684	$789	$884	$941	$1,049	$1,182
3	Annual fixed charges on cost of pipe, insulation, etc.	$309	$388	$484	$585	$692	$827
4	Annual cost of heat lost by radiation	$86.46	$105.44	$121.40	$138.77	$153.52	$160.79
5	Pressure loss in header, lb. per sq. in.	8.05	2.19	0.82	0.33	0.16	0.12
6	Inside diameter of pipe, inches.	7.625	9.75	11.75	14.00	16.00	17.00
7	Inside area of pipe, sq. ft.	0.317	0.518	0.751	1.07	1.39	1.57
8	Velocity, ft. per minute.	11,950	7,300	5,040	3,540	2,720	2,410
9	Velocity head in header, lb. per sq. in.	1.94	0.725	0.345	0.17	0.10	0.07
10	Total pressure loss from boiler drum to turbine, lb. per sq. in.	16.99	9.92	8.16	7.50	7.26	7.13
11	Pressure of steam at turbine nozzle, lb. per sq. in. ga.	233.	240.	242.	243.	243.	243.
12	Radiation from header, thous. Btu. per hr.	60.2	72.6	83.5	95.5	105.5	110.5
13	Total heat of steam at turbine nozzle, Btu. per lb.	1,289.8	1,289.7	1,289.6	1,289.5	1,289.4	1,289.3
14	Superheat at turbine nozzle, deg. F.	152.	150.	150.	149.	149.	149.
15	Water rate of turbine, lb. steam per kw. hr.	13.14	13.057	13.04	13.035	13.039	13.04
16	Annual cost of fuel to operate turbine	$124,323	$123,527	$123,355	$123,306	$123,323	$123,321
17	Total annual cost	$124,718	$124,020	$123,960	$124,033	$124,169	$124,309



— FIG. 3. —

— GRAPHIC STUDY OF HEADER SIZE DETERMINATION. —

$V = 2.20$
F is obtained from col. 6, Table 8, Appendix IV.
then:

$$p = (1720)^2 \times 165 \times 2.2 \times F = 1.075 \times 10^9 \times F.$$

Item 6.

From "National Pipe Standards", table p. 649, col. 3.

Item 7.

$$\text{Area, sq. ft.} = (\text{item } 6)^2 \times \frac{0.7854}{144} = 0.00545\ (\text{item } 6)^2$$

Item 8.

$$\text{Velocity (ft. per min.)} = \frac{\text{lb. steam per min.} \times \text{sp. vol.}}{\text{area in sq. ft.}} =$$

$$\frac{1720 \times 2.2}{\text{area}} = \frac{3784}{\text{item } 7}.$$

Item 9.

$$\text{Velocity head, lb. per sq. in.} = \frac{\dfrac{(\text{ft. per sec.})^2}{2\,g}}{\text{sp. vol.} \times 144} =$$

$$\frac{(\text{ft. per min.})^2}{3600 \times 2 \times 32.2 \times \text{sp. vol.} \times 144} = \frac{(\text{item } 8)^2}{73{,}500{,}000}.$$

Item 10.

The pressure loss from saturated steam drum to header, including dry pipe, superheater, non-return valve, feeder, and all intermediate fittings is 7 lb. per sq. in. (From test data on similar boiler, corrected to the proper rating).

Item 10 = item 5 + item 9 + 7.

Item 11.

Item 11 = 250 - item 10.

Item 12.

Btu. per hr. = length (202 ft.) x item 1* x (1 - 0.01 item 3*)
*Col. 3, Table 1.

Item 13.

Total heat of steam at boiler nozzle (265 lb. per sq. in. abs. and 150° superheat) = 1290.4

$$\text{Total heat of steam at turbine nozzle} = 1290.4 - \frac{\text{item } 12}{\text{lb. steam per hr.}}$$

Item 14.

Obtained by interpolation from F. O. Ellenwood's "Steam Charts", using item 11 and item 13 as data.

Note that the higher superheat for small pipes than for large pipes is due to two causes: (1) less radiation, and (2) throttling (expansion at constant total heat), and that for 8-inch pipe there is more superheat at the turbine than at the boiler, even though the total heat is lower at the turbine due to radiation.

Item 15.

The water rate of the turbine at the design load and steam conditions is given in the guarantee as 12.9 lb. per kwh. With steam at 250 lb. gage and 150 deg. superheat (specifications) the electrical energy obtained from one pound of steam is $\frac{3412}{12.9} = 264.50$ Btu. per pound.

A study of the pressure corrections used by the Westinghouse Electric and Mfg. Company and a check calculation by thermodynamics indicate that the loss of available energy amounts to 0.25 Btu. per pound of steam for each pound decrease of pressure, within the range of this problem.

The heat lost by radiation has a direct effect by reducing the available energy per pound of steam by an amount equal to the total Btu. lost by radiation per hour, divided by the total weight of steam flowing per hour.

The net electrical energy available per pound of steam then is 264.50 minus the losses due to pressure drop and radiation, and the corrected water rate of the turbine is equal to 3412 divided by the net available energy per pound of steam.

Item 16.

Heat carried by one lb. steam from fuel to turbine nozzle = item 13 - feed water temp. (i. e. 210°) + 32 = item 13 - 178.

$$\text{Item 16} = \frac{\overset{\text{kw.}}{8000} \times \overset{\text{hr.}}{6400} \times \overset{\text{w. r.}}{\text{item 15}} \times \overset{\text{Btu. steam}}{(\text{item 13} - 178)} \times \overset{\text{Coal}}{3.50}}{\underset{\text{Coal}}{13{,}500} \times \underset{\text{eff}}{0.78} \times \underset{\text{ton}}{2000}} =$$

8.51 x item 15 (item 13 - 178).

Item 17.

Item 17 = item 3 + item 4 + item 16.

20 - CONCLUSIONS ON HEADER SIZE.

An inspection of the total cost curve of Fig. 3 and item 17 of Table No. 2 indicates that a 12-inch header is the economical size to install. It is noted further that any of the six sizes considered would involve an annual loss of less than $800.00 as compared with the economical size. This is true because the disadvantages of increased cost and increased radiation loss for the larger

sizes are nearly counterbalanced by the advantage of less pressure loss. It is not to be concluded, however, that this condition occurs in all cases and that any size within a wide range will constitute an economical selection. (See conclusions at end of Appendix II).

APPENDIX II.

DESIGN OF A COMPLICATED STEAM HEADER.

21 - DESCRIPTION. (See Fig. 4)

The plant consists of four 15,000 kw. (80% power factor) turbo-generators served by eight 1500 hp. boilers. A 1500 kw. d. c. turbo-generator, and a motor-generator set receiving power from the main station generators, supply auxiliary power, the two machines providing a flexible link for the manipulation of exhaust steam to effect a station heat balance. One turbine driven feed pump and steam jet air exhausters receive steam from an auxiliary header located under the main header under the generator room.

22 - SELECTION OF INSULATION.

The steam conditions are the same as in Appendix I, and with the same coal cost, colorific value, and boiler efficiency, the insulation specification will be the same, viz., 2-inches of Asbesto-Sponge-Felted for all sizes. (See par. heading #14ff.)

23 - LOAD CONDITIONS.

The anticipated load is an industrial load consisting of several steel mills connected electrically by a super-power system. The normal design load is 45,000 kw. during 6400 hours per year.

24 - STEAM REQUIREMENTS.

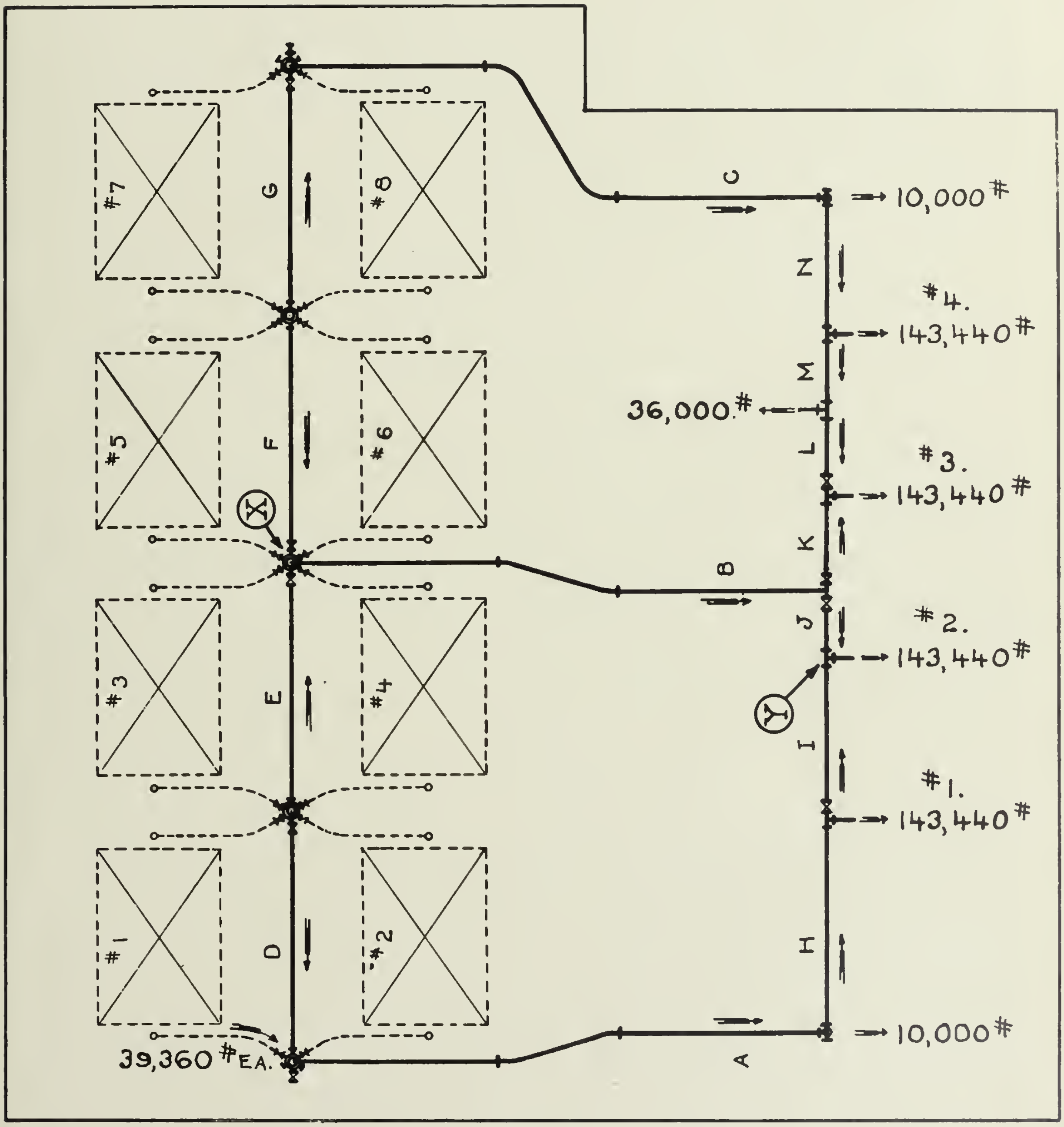
The steam requirements for the plant (excluding intermittent demands such as soot-blowers, ash dumps, etc.) under design load conditions are:

Item	Equipment	Lb. steam per hr.
a	Main generators 45,000 kw. @ 12.75 lb.	573,750
b	Auxiliary generator 1200 kw. @ 30 lb.	36,000
c	Other steam auxiliaries	20,000
	Total	629,750

The main generator load would be normally carried on three machines, but as any one of the four might be off the line, it is assumed for design purposes that each machine carries one fourth the total load. Similarly, seven boilers would normally carry the load, but it is assumed that each carries one eighth of the total requirement. The actual steam distribution in the header system would be different for each possible combination of units in service, and it is quite certain that the assumptions made closely represent <u>average</u> conditions.

The steam required by each turbine =

$$\frac{45{,}000 \times 12.75}{4} = 143{,}440 \text{ lb. per hr.}$$



—FIG. 4.—

—LAYOUT OF A COMPLICATED STEAM HEADER.—

It is assumed that one-half of the 20,000 lb. per hr. used by miscellaneous auxiliaries is taken from each end of the main header under the generator room. The steam distribution in the header is discussed under item 5, par. heading # 27 .

25 - PROVISION FOR FUTURE EXTENSION.

The possibility of future extension must not be overlooked. In a layout like Figure 4, however, extension may be disregarded, since additional generators would be accompanied by additional boilers, and the piping would be extended with additional cross-branches, the system thus expanding similarly to the "Unit System", and each succeeding unit possessing characteristics similar to the original unit.

26 - SELECTION OF HEADER SIZE.

Table No. 3 is the tabular solution of the header size problem. A complete set of explanatory notes follows the table.

Fig. 5 is a graphic presentation of the important items of the table. The curves are numbered to correspond with the items they represent.

A study of Fig. 5, (or Fig. 3) is enlightening. It is noted that curve 17 is a "U"-curve. Its components are curves 3, 4, and 16. Curves 3 and 4 are increasing functions, i. e. they increase as the pipe size increases. Radiation increases because of the greater amount of surface exposed, and the fixed charges increase because of the higher cost of materials. Curve 16 is a decreasing function because the water rate of the generators (curve 15) decreases due to the lesser pressure loss in large pipes (curve 10). The sum of an increasing and a decreasing function always results in a "U"-curve, which must have a minimum value. The finding of this minimum value by analytical considerations is the basis of the "Rational" method of design.

27 - EXPLANATION OF ITEMS IN TABLE NO. 3.

Item 1.

From quotation submitted by the Pittsburgh Valve Foundry and Construction Company, March, 1922. (Correction made for header length).

Item 2.

From quotation submitted by H. W. Johns-Manville Company, dated February 17, 1922. (Correction made for header length).

Item 3.

Item 3 = 0.15 (item 1 + item 2). The fixed charges are assumed to be: interest 6%, depreciation 5%, repairs and maintenance 1%, taxes, insurance, etc. 3%. Total 15%.

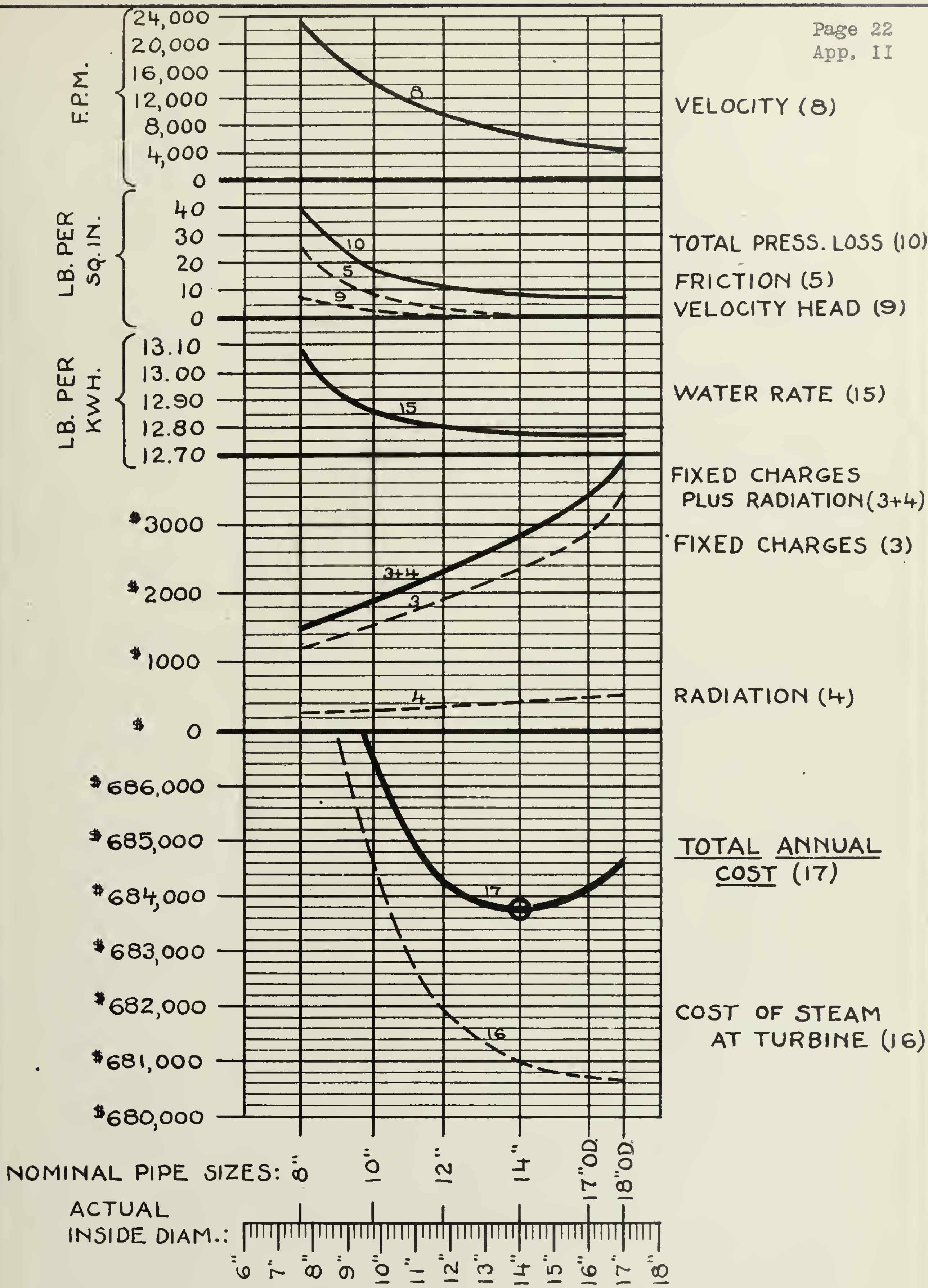
Item 4.

Cost of heat lost = length of header (i. e. 632 ft.) x item 4, column 3, Table 1, Appendix I.

Table No. 3

Determination of Economical Header Size.

Item Number	Nominal Size of Pipe, Inches.	8	10	12	14	17 O.D.	18 O.D.
	Column Numbers.	1	2	3	4	5	6
1	Cost of header, incl. fittings, erected.	$6,030	$7,920.	$10,170	$12,800	$16,050	$19,840
2	Cost of insulation, incl. fittings, placed	$2,240	$2,587.	$2,882	$3,200	$3,530	$3,860
3	Annual fixed charges on cost of pipe insulation, etc.	$1,242	$1,576	$1,960	$2,400	$2,938	$3,555
4	Annual cost of heat lost by radiation	$271	$330	$380	$434	$480	$503
5	Pressure loss in header, lb. per sq. in.	25.9	7.0	2.6	1.06	0.53	0.39
6	Inside diameter of pipe, inches	7.625	9.750	11.75	14.00	16.00	17.00
7	Inside area of pipe, sq. ft.	0.317	0.518	0.751	1.07	1.39	1.57
8	Velocity, ft. per min. (in sections A, B, and C)	22,800	14,000	9,650	6,760	5,210	4,610
9	Velocity head, lb. per sq. in.	7.07	2.67	1.27	0.82	0.37	0.29
10	Total pressure loss from boiler drum to turbine, lb. per sq. in.	39.97	16.67	10.87	8.68	7.90	7.68
11	Pressure of steam at turbine nozzle, lb. per sq. in ga.	210.03	233.33	239.13	241.32	242.10	242.32
12	Radiation from header, thous. Btu. per hr.	188.3	227.0	261.0	299.0	330.0	345.5
13	Total heat of steam at turbine nozzle, Btu. per lb.	1,290.1	1,290.0	1,290.0	1,289.9	1,289.9	1,289.8
14	Superheat at turbine nozzle, deg. F.	159°	153°	151°	151°	151°	151°
15	Water rate of turbine, lb. per kw. hr.	13.081	12.851	12.801	12.785	12.781	12.781
16	Annual cost of fuel to operate turbine	$696,820	$684,506	$681,843	$680,929	$680,716	$680,655
17	Total annual cost	$698,332	$686,412	$684,183	$683,763	$684,134	$684,713



—FIG. 5.—

—GRAPHIC STUDY OF HEADER SIZE DETERMINATION.—

Item 5.

In order to determine the pressure loss in a complicated header, it is necessary to determine the distribution of steam in the various parts of the header. Figure 4 and Table No. 4 demonstrate the method of calculation. Each section of the header is given a designating letter, which appears in column 1 of the table (4).

In column 2 the length of each section is indicated.

In column 3 the weight of steam flowing in each section is listed. This distribution is based upon the assumption that all boilers are on the line, and the first trial assumes that the three cross branches, A, B, and C, carry equal flow. The flow from each boiler outlet is equal to the total steam flow divided by the number of outlets, = 629,750 ÷ 16 = 39,360 lb. per hr. The flow in each of the cross branches = 629,750 ÷ 3 = 209,920 lb. per hr. Column 3 can now be filled in by starting with any convenient branch, as A, and adding or subtracting, as the case may be, as steam is fed into or bled from the header. Column 4 need not be filled in unless the first trial is found to be satisfactory.

Column 5 is obtained by applying the formula method, as in item 5, Table No. 2.

Now, if the steam distribution as assumed is correct, the system will be in dynamic equilibrium, and the pressure loss between any two points, such as X and Y, figure 4 will be the same, whether the path be taken via A, B, or C.

The pressure loss from X to Y via A for 14-inch pipe = D + A + H + I - E = 1.265 lb. per sq. in. Via B = B + J = 0.872 lb. per sq. in. Via C = G + C + N + M + L + J - F - K = 1.212. Comparison of these three losses shows that the flow through B was assumed too low as the loss by that route is lower than by A or C. The average of these losses is 1.116 lb. per sq. in.

Second trial: To approximate a corrected flow for A, B, and C, the flow in each of these branches is multiplied by the square root of 1.116 and divided by the square root of the first trial loss.

Columns 6 and 8 constitute the second trial. The calculation is the same as for columns 3 and 5.

The loss from X to Y with the values tabulated in column 8 is: via A 1.095, via B 1.075, via C 0.973. The average is 1.048. There is still some variance between these values, but a third trial is not advisable, since there is a variation between the pressure drops to the various turbines anyway, hence the average of the three values, or 1.048 is taken as the drop to the point Y, or turbine #2.

Column 7 is calculated by:

$$\text{Vel., ft. per min.} = \frac{\text{lb. per hr. x sp. vol.}}{\text{60 x area, sq. ft.}} = 0.0343 \text{ x Col. 6.}$$

i

Table No.4
Pressure Loss Analysis of Fig. 4.
For 14-Inch Header.

		First Trial			Second Trial		
Section	Length Feet	Lbs. per hour	Velocity f.p.m.	Pressure loss	Lbs. per hour	Velocity f.p.m.	Pressure loss
1	2	3	4	5	6	7	8
A	102	209,920		0.840	197,325	6,770	0.743
B	103	209,920		0.849	235,110	8,060	1.055
C	111	209,920		0.915	197,325	6,770	0.808
D	43	131,200		0.139	118,605	3,720	0.114
E	43	26,240		0.006	38,835	1,333	0.012
F	43	26,240		0.006	38,835	1,333	0.012
G	43	131,200		0.139	118,605	3,720	0.114
H	37	199,920		0.276	187,325	6,440	0.240
I	28	56,480		0.016	43,885	1,504	0.010
J	16	86,960		0.023	99,555	3,420	0.020
K	16	122,960		0.045	135,555	4,650	0.055
L	12	20,480		0.001	7,885	270	0.000+
M	12	56,480		0.007	43,885	1,504	0.004
N	24	199,920		0.178	187,325	6,440	0.094

Since the pressure drop to turbine #2 is 1.048 lb. per square inch, the drop to other turbines may be obtained by adding or subtracting the losses in sections between the tee for turbine #2 and the tees for the other turbines. The losses to the various turbines are found to be: #1 - 1.038, #2 - 1.048, #3 - 1.083, #4 - 1.079. The average is 1.062, which is entered in column 4 of Table No. 3 for item 5. The pressure loss for the other sizes is calculated by multiplying 1.062 by the ratio of the respective values of factor F from Table No. 8, Appendix IV.

Item 6.

From "National Pipe Standards", table p. 649, col. 3.

Item 7.

$$\text{Area, sq. ft.} = (\text{item } 6)^2 \times \frac{0.7854}{144} = 0.00545\ (\text{item } 6)^2$$

Item 8.

Velocity (ft. per min.) in interconnections, A, B, and C, =

$$\frac{\text{lb. steam per min. x sp. vol.}}{\text{Area in sq. ft.}} = \frac{9870 \times 2.2}{3 \times \text{area}} = \frac{7{,}238}{\text{item } 7}$$

Item 9.

$$\text{Velocity head, lb. per sq. in.} = \frac{\dfrac{(\text{ft. per sec.})^2}{2\ g}}{\text{sp. vol.} \times 144} =$$

$$\frac{(\text{ft. per min.})^2}{3600 \times 2 \times 32.2 \times \text{sp. vol.} \times 144} = \frac{(\text{item } 8)^2}{73{,}500{,}000}$$

Item 10.

The pressure loss from saturated steam drum to header, including dry pipe, superheater, non-return valve, feeder, and all intermediate fittings is 7 lb. per sq. in. (From test data on similar boiler, corrected to the proper rating).

Item 10 = item 5 + item 9 + 7.

Item 11.

Item 11 = 250 - item 10.

Item 12.

Btu. per hr. = length (632 ft.) x item 1* (1 - 0.01 item 3*).
*Col. 3, Table 1.

Item 13.

Total heat of steam at boiler nozzle (265 lb. per sq. in. Abs. and 150° superheat) = 1290.4

$$\text{Total heat of steam at turbine nozzle} = 1290.4 - \frac{\text{item 12}}{\text{lb. steam per hr.}}$$

Item 14.

Obtained by interpolation from F. O. Ellenwood's "Steam Charts", using item 11 and item 13 as data.

Note that the higher superheat for small pipes than for large pipes is due to two causes: (1) less radiation, and (2) throttling (expansion at constant total heat), and that for all sizes of pipe there is more superheat at the turbine than at the boiler, even though the total heat is lower at the turbine due to radiation.

Item 15.

The water rate of the turbine at the design load and steam conditions is given in the guarantee as 12.75 lb. per kwh. With steam at 250 lb. gage and 100 deg. superheat (specifications) the electrical energy obtained from one pound of steam is $\frac{3412}{12.75} = 268$ Btu. per pound.

A study of the pressure corrections used by the Westinghouse Electric and Mfg. Company and a check calculation by thermodynamics indicate that the loss of available energy amounts to 0.25 Btu. per pound of steam for each pound decrease of pressure, within the range of this problem.

The heat lost by radiation has a direct effect by reducing the available energy per pound of steam by an amount equal to the total Btu. lost by radiation per hour, divided by the total weight of steam flowing per hour.

The net electrical energy available per pound of steam, then, is 268 minus the losses due to pressure drop and radiation, and the corrected water rate of the turbine is equal to 3412 divided by the net available energy per pound of steam.

Item 16.

Heat carried by one pound of steam from fuel to turbine nozzle = item 13 - feed water temp. (i. e. 210°) + 32 = item 13 - 178.

$$\text{Item 16} = \frac{\overset{\text{kw}}{45{,}000} \times \overset{\text{hr.}}{6400} \times \overset{\text{w. r.}}{\text{item 15}} \times \overset{\text{Btu. steam}}{(\text{item 13}-178)} \times \overset{\text{Coal}}{3.50}}{\underset{\text{Coal}}{13{,}500} \times \underset{\text{eff.}}{0.78} \times \underset{\text{ton}}{2000}} =$$

$47.9 \times$ item 15 $\times$ (item 13 - 178).

Item 17.

Item 17 = item 3 + item 4 + item 16.

28 - CONCLUSIONS ON HEADER SIZE.

Inspection of the total cost curve of Fig. 5 and item 17 of Table No. 3 indicates that a 14-inch header is the economical size to install.

29 - CLOSING DISCUSSION OF THE RATIONAL METHOD.

The failure of the velocity method is proved by a comparison of Tables

Nos. 2 and 3. In table No. 2 it is found that the velocity corresponding to the economical size is 5000 ft. per min. Now, if the second design problem (Table No. 3) were solved on the basis of 5000 ft. per min. velocity in the main branches, A, B, and C, a 17-inch (O. D.) header would have been selected. Item 17 indicates that the annual charges against this header would be $350.00 per year greater than those against the 14-inch. Hence, if a designer's time is required for one month (which is much longer than should be required) to make a rational analysis, his salary would be saved in from one to two years, which is considered a good investment. Further, in col. 7 of Table No. 4 it is found that velocities in the 14-inch header range from 270 to 8060 ft. per min. Thus the velocity method loses most of its significance at once.

It is interesting to note that the total pressure loss for the economical headers in the two problems are 8.16 and 8.68 lb. per sq. in. respectively. It does not follow, however, that 8 to 9 lb. may be assumed as the economical drop for all cases. Two examples will serve to point out the fallacy of such a conclusion. (1) There are many factors which affect the problem, one of which is fuel cost. In some localities fuel costs are double those in other localities, and where the cost is high, larger headers are warranted since the energy lost through pressure drop has a greater value than with low fuel cost. The effect of high fuel cost on Fig. 5 is to raise curve 16, which tends to shift the minimum value of curve 17 towards the right. (2) It is obvious that a greater pressure drop is warranted in a stand-by plant operating with a low yearly load factor than in a high load factor plant, since in the case of the former the financial cost of an "excessive" pressure drop for only a few hours a day or month is small, and the high fixed charges on the greater cost of a large header are not warranted.

A well known Central Station designer has criticised the rational method, claiming that too much time is required, that a designer who is proficient enough to do such work is worth much more on other phases of the work, and that satisfactory results are obtained by a good guess based upon experience. This criticism may have some justification, but it is the author's firm contention that even if the pressure of time does require "a good guess" rather than complete analysis, that a much more intelligent guess may be made if the designer has gone through at least one complete problem by the rational method, which will endow him with a knowledge of the various factors affecting the problem, and their relative importance.

Further, for a designer who proposes to use the pressure loss method of design, the additional work required to complete the rational method is not great, and it will undoubtedly be found to be justified.

In view of the facts brought out in this study, the rational method is recommended by the author as the most satisfactory method of steam header design.

APPENDIX III.

RADIATION LOSSES FROM BARE AND INSULATED PIPES.

Probably the most reliable experimental data available on radiation losses, are those reported by Mr. L. B. McMillan in Vol. 37 of Trans. A.S.M.E.

Table No. 5, published in "Johns-Manville Service to Power Users" is based upon McMillan's experimental data, and is considered reliable. The temperature differences at the column headings in this table refer to those between outside pipe surface and surrounding air. Analysis of McMillan's experiments indicates that with zero velocity inside the pipe, the inner surface and metal resistance for bare pipes is about 1.27 percent of the total resistance for saturated steam, and 9.45 percent for superheated steam. The magnitude of the inner surface resistance is known to decrease as velocity increases, but the exact relation between these two quantities is not known. It is known, however, from tests made with air, that a small increase in velocity is accompanied by a large decrease in surface resistance. It may therefore be concluded that at the customary high velocities in steam headers, the inner surface resistance is less than one or two percent of the total resistance, and hence sufficient accuracy is obtained by neglecting the resistances imposed by the inner surface and the pipe metal. Table No. 5 may then be used by interpreting the column headings as "temperature difference between steam and surrounding air".

The efficiency of an insulating covering is defined as the ratio of the amount of heat saved by using the insulation, to the amount which would be lost if the pipe were left bare, expressed as a percentage.

To determine the amount of heat radiated from an insulated pipe, then, it is necessary to determine the amount radiated from a bare pipe, and multiply this loss by the efficiency of the insulation.

The efficiencies of commercial pipe insulating materials vary from about 50 percent to about 97 percent, depending upon the nature of the material, the thickness of the insulation, the temperature difference, and the pipe size. Table No. 6 taken from "Johns-Manville Service to Power Users" shows the efficiencies of Johns-Manville "Asbesto-Sponge-Felted" insulation.

Table No. 7, from "Johns-Manville Service to Power Users" gives the list prices of all classes of pipe coverings. Discounts vary with the grade of material and market value.

TABLE NO. 5.

RADIATION LOSS FROM BARE PIPE.

Total Heat Loss in B. t. u. Per Hour Per Lineal Foot of Bare Pipe of Different Sizes and Per Square Foot of Flat Surfaces and at Various Temperature Differences

(For finding losses at temperatures between those shown, the B. t. u. Differences per Degree are given in small type between the Main Columns)

Pipe Size	Area of Pipe Surface per lin. ft.	50°		100°		150°		200°		250°		300°		350°		400°		450°		500°
		Heat Loss in B. t. u. per lineal ft. per Hour																		
½″	.220	21.5	*.52	47.3	*.64	79.2	*.76	117.3	*.90	162.3	*1.06	215.2	*1.28	279.1	*1.52	355.1	*1.93	451.4	*2.37	569.8
¾″	.274	26.8	.64	59.0	.79	98.6	.96	146.8	1.11	202.1	1.33	268.5	1.58	347.6	1.89	442.2	2.40	562.2	2.95	709.7
1″	.344	33.6	.81	74.0	1.00	123.8	1.19	183.4	1.41	253.7	1.67	337.4	1.98	436.5	2.37	555.2	3.03	705.4	3.69	891.
1¼″	.435	42.5	1.01	93.6	1.26	156.6	1.51	231.9	1.78	320.8	2.09	425.4	2.53	551.9	3.00	702.1	3.80	892.6	4.68	1126.7
1½″	.498	48.7	1.17	107.2	1.44	179.3	1.72	265.4	2.04	367.3	2.39	487.	2.90	631.8	3.44	803.8	4.36	1021.9	5.36	1289.8
2″	.622	60.9	1.46	133.9	1.80	223.9	2.15	331.5	2.54	458.7	2.99	608.3	3.62	789.2	4.29	1003.9	5.45	1276.3	6.69	1611.
2½″	.753	73.4	1.76	161.6	2.18	270.4	2.60	400.3	3.07	553.9	3.61	734.5	4.37	952.8	5.19	1212.1	6.58	1541.1	8.08	1945.1
3″	.917	89.6	2.15	197.3	2.66	330.1	3.17	488.8	3.75	676.3	4.41	896.8	5.33	1163.4	6.33	1480.	8.03	1881.7	9.87	2375.
3½″	1.047	102.3	2.46	225.3	3.03	376.9	3.62	558.1	4.28	772.2	5.04	1024.	6.09	1328.4	7.23	1689.9	9.17	2148.4	11.3	2711.7
4″	1.178	115.1	2.77	253.5	3.41	424.2	4.07	627.9	4.82	868.8	5.67	1152.1	6.85	1494.6	8.13	1901.3	10.3	2417.3	12.7	3051.
4½″	1.309	127.9	3.07	281.5	3.79	470.9	4.53	697.2	5.35	964.7	6.29	1279.2	7.61	1659.5	9.03	2111.1	11.05	2684.	14.1	3387.7
5″	1.456	142.2	3.42	313.1	4.21	523.8	5.03	775.5	5.95	1073.	7.00	1423.	8.46	1846.	10.0	2348.4	12.7	2985.7	15.7	3768.5
6″	1.734	169.4	4.05	371.9	5.04	623.9	6.00	923.7	7.09	1278.1	8.34	1694.9	10.1	2198.7	12.0	2797.1	15.2	3556.2	18.6	4488.5
7″	2.00	195.0	4.71	430.4	5.79	720.0	6.92	1066.0	8.19	1475.6	9.61	1956.	11.66	2539.	13.78	3228.	17.46	4101.	21.6	5180.
8″	2.257	220.6	5.30	485.7	6.54	812.5	7.81	1203.	9.23	1664.5	10.8	2207.3	13.1	2863.6	15.6	3642.8	19.8	4631.4	24.3	5845.6
9″	2.52	246.0	5.92	542.	7.30	907.	8.72	1343.	10.34	1860.	12.1	2465.	14.7	3200.	17.4	4070.	22.0	5170.	27.2	6530.
10″	2.817	275.4	6.62	606.2	8.16	1014.1	9.75	1501.5	11.5	2077.5	13.6	2755.	16.4	3574.1	19.5	4546.6	24.7	5780.5	30.3	7296.
11″	3.08	300.	7.26	663.	8.92	1109.	10.66	1642.	12.6	2272.	14.76	3010.	17.9	3905.	21.3	4972.	26.9	6315.	33.3	7980.
12″	3.34	326.	7.86	719.	9.68	1203.	11.54	1780.	13.7	2465.	16.02	3266.	19.4	4235.	23.1	5390.	29.2	6850.	36.0	8650.
14″o. d.	3.66	357.	8.59	786.	10.64	1318.	12.64	1950.	15.0	2700.	17.06	3580.	21.3	4645.	25.2	5905.	31.9	7500.	39.5	9475.
16″o. d.	4.19	408.	9.84	901.	12.2	1510.	14.5	2233.	17.2	3095.	20.1	4100.	24.4	5320.	28.9	6765.	36.5	8590.	45.2	10850.

Flat, Curved, or Cylindrical Surfaces.	50°		100°		150°		200°		250°		300°		350°		400°		450°		500°
Heat Loss in B. t. u. per sq. ft. per Hour	97.5	2.35	215.2	2.90	360.0	3.46	533.0	4.10	737.8	4.80	978.0	5.83	1269.4	6.89	1614.0	8.73	2050.6	10.8	2590.0
Heat Loss in B. t. u. per sq. ft. per degree temperature difference per Hour	1.950		2.152		2.400		2.665		2.951		3.260		3.627		4.035		4.557		5.180

*Example 2″ Pipe, 235° Temp. Difference. 235° − 200° = 35°; 35° x 2.54 (B. t. u. per degree) = 88.9 B. t. u. 331.5 + 88.9 = 420.4; B. t. u. loss at 235° Temp. difference.

TABLE NO. 6.

EFFICIENCY OF ASBESTO-SPONGE-FELTED SECTIONAL INSULATION.

Efficiencies of Standard Thick Johns-Manville Asbesto-Sponge Felted Sectional Pipe Insulation on Various Sizes of Pipes

Pipe Size, Inches	Temperature Difference Between Steam in Pipe and Air Surrounding Pipe									
	50°	100°	150°	200°	250°	300°	350°	400°	450°	500
	Per Cent Efficiencies									
½	68.5%	71.2%	73.3%	75.5%	77.1%	78.7%	80.4%	81.9%	83.1%	84.5%
¾	71.9	74.3	76.2	78.1	79.6	81.	82.6	83.8	85.	86.2
1	74.3	76.5	78.2	79.9	81.3	82.6	84.	85.2	86.2	87.4
1¼	75.7	77.7	79.5	81.	82.3	83.5	84.9	86.	86.9	88.
1½	77.	79.	80.5	82.	83.2	84.4	85.7	86.7	87.6	88.6
2	78.6	80.4	81.9	83.3	84.4	85.5	86.7	87.6	88.5	89.3
2½	79.8	81.5	82.9	84.3	85.3	86.3	87.5	88.4	89.2	90.3
3	80.6	82.2	83.6	84.9	85.9	86.8	87.9	88.8	89.6	90.3
3½	81.2	82.8	84.1	85.4	86.3	87.3	88.3	89.2	89.9	90.6
4	81.8	83.3	84.5	85.8	86.7	87.6	88.7	89.5	90.2	90.9
4½	82.1	83.6	84.8	86.	87.	87.9	88.9	89.7	90.4	91.1
5	82.3	83.8	85.	86.2	87.1	88.	89.	89.8	90.5	91.2
6	82.7	84.2	85.4	86.5	87.4	88.3	89.3	90.1	90.7	91.4
7	83.	84.5	85.6	86.8	87.6	88.5	89.5	90.2	90.9	91.6
8	83.4	84.8	85.9	87.	87.9	88.7	89.7	90.4	91.1	91.7
9	83.5	84.9	86.	87.2	88.	88.8	89.8	90.5	91.2	91.8
10	83.8	85.1	86.2	87.3	88.2	89.	89.9	90.6	91.3	91.9

Efficiencies of 1½″ Thick Johns-Manville Asbesto-Sponge Felted Sectional Pipe Insulation on Various Sizes of Pipe

Pipe Size, Inches	Temperature Difference Between Steam in Pipe and Air Surrounding Pipe									
	50°	100°	150°	200°	250°	300°	350°	400°	450°	500°
	Per Cent Efficiencies									
½	70.3%	72.9%	75. %	76.8%	78.4%	80. %	81.5%	82.9%	84.2%	85.2%
¾	73.7	75.9	77.7	79.4	80.9	82.4	83.5	84.9	86.	87.
1	76.1	78.1	79.8	81.3	82.7	84.1	85.1	86.3	87.3	88.1
1¼	77.8	79.7	81.2	82.5	83.8	85.1	86.1	87.2	88.2	89
1½	79.	80.7	82.3	83.5	84.7	86.	86.9	88.	88.8	89.9
2	81.	82.7	84.	85.2	86.2	87.2	88.1	89.1	89.9	90.6
2½	82.1	83.7	85.1	86.1	87.	88.1	88.8	89.8	90.5	91.1
3	83.	84.5	85.7	86.8	87.7	88.7	89.4	90.3	91.	91.6
3½	83.6	85.	86.2	87.2	88.1	89.1	89.8	90.6	91.3	91.9
4	84.2	85.5	86.7	87.7	88.5	89.5	90.1	91.	96.	92.1
4½	84.6	85.9	87.1	88.	88.8	89.8	90.4	91.2	91.8	92.3
5	85.	86.3	87.4	88.3	89.1	90.	90.6	91.4	91.2	92.5
6	85.4	86.7	87.8	88.6	89.4	90.3	90.9	91.7	92.2	92.7
7	85.9	87.1	88.1	89.	89.8	90.6	91.2	91.9	92.5	92.9
8	86.2	87.5	88.4	89.1	90.	90.8	91.4	92.1	92.7	93.1
9	86.4	87.7	88.6	89.3	90.1	90.9	91.5	92.2	92.8	93.2
10	86.6	87.8	88.8	89.6	90.3	91.	91.6	92.3	92.9	93.4

TABLE NO. 6 - (CONT'D)

Efficiencies of 2″ Thick Johns-Manville Asbesto-Sponge Felted Sectional Pipe Insulation on Various Sizes of Pipe

Temperature Difference Between Steam in Pipe and Air Surrounding Pipe

Pipe Size, Inches	50°	100°	150°	200°	250°	300°	350°	400°	450°	500°
	Per Cent Efficiencies									
½	73.2%	75.6%	77.5%	79.2%	80.7%	82. %	83.5%	84.6%	86.1%	87.4%
¾	76.5	78.6	80.3	81.8	83.1	84.2	85.5	86.5	87.8	88.8
1	78.8	80.7	82.2	83.6	84.8	85.8	87.	87.8	89.	90.
1¼	80.5	82.	83.6	84.9	86.	87.1	88.	88.8	89.8	90.8
1½	81.7	83.4	84.7	85.9	86.9	87.8	88.7	89.5	90.4	91.2
2	83.6	85.1	86.2	87.2	88.1	89.	89.9	90.6	91.4	92.1
2½	84.3	86.1	87.2	88.2	89.	89.7	90.5	91.2	92.	92.7
3	85.6	86.8	87.9	88.8	89.6	90.3	91.1	91.7	92.5	93.2
3½	86.2	87.4	88.4	89.3	90.1	90.8	91.5	92.1	92.8	93.5
4	86.7	87.9	88.8	89.7	90.5	91.1	91.8	92.4	93.1	93.7
4½	87.1	88.3	89.2	90.	90.7	91.3	92.1	92.6	93.3	93.9
5	87.5	88.6	89.5	90.3	91.	91.6	92.3	92.8	93.5	94.1
6	88.	89.1	89.9	90.7	91.3	91.9	92.6	93.1	93.8	94.3
7	88.4	89.4	90.2	91.	91.6	92.2	92.8	93.3	94.	94.5
8	88.6	89.6	90.4	91.2	91.8	92.4	93.	93.4	94.1	94.6
9	88.9	89.9	90.6	91.4	92.	92.5	93.1	93.6	94.2	94.7
10	89.1	90.1	90.8	91.5	92.2	92.7	93.3	93.7	94.3	94.8

Efficiencies of 2½″ Thick Johns-Manville Asbesto-Sponge Felted Sectional Pipe Insulation on Various Sizes of Pipe

Temperature Difference Between Steam in Pipe and Air Surrounding Pipe

Pipe Size, Inches	50°	100°	150°	200°	250°	300°	350°	400°	450°	500°
	Per Cent Efficiencies									
½	75. %	77.3%	78.9%	80.7%	82.1%	83.7%	84.8%	85.9%	87. %	88.%
¾	78.2	80.	81.6	83.2	84.4	85.7	86.7	87.7	88.5	89.3
1	80.5	82.2	83.5	85.	86.	87.2	88.1	89.	89.7	90.4
1¼	82.2	83.8	84.9	86.2	87.2	88.3	89.1	90.	90.6	91.3
1½	83.4	84.9	86.	87.2	88.1	89.1	89.9	90.6	91.2	91.9
2	85.2	86.5	87.5	88.6	89.4	90.3	91.	91.7	92.2	92.8
2½	86.3	87.5	88.4	89.4	90.2	91.	91.7	92.3	92.8	93.3
3	87.1	88.3	89.1	90.1	90.8	91.6	92.2	92.7	93.2	93.7
3½	87.8	88.9	89.7	90.6	91.2	92.	92.6	93.1	93.5	94.
4	88.3	89.3	90.1	90.9	91.6	92.3	92.9	93.4	93.8	94.3
4½	88.7	89.7	90.4	91.2	91.9	92.6	93.1	93.6	94.	94.5
5	89.	90.	90.7	91.5	92.	92.8	93.3	93.8	94.2	94.6
6	89.5	90.4	91.1	91.9	92.	93.1	93.6	94.1	94.5	94.8
7	89.9	90.8	91.4	92.2	92.7	93.4	93.8	94.3	94.7	95.
8	90.1	91.	91.6	92.4	92.9	93.5	94.	94.4	94.8	95.1
9	90.3	91.2	91.8	92.5	93.1	93.7	94.1	94.6	94.9	95.2
10	90.5	91.4	92.	92.7	93.2	93.8	94.2	94.7	95.	95.3

Efficiencies of 3″ Thick Johns-Manville Asbesto-Sponge Felted Sectional Pipe Insulation on Various Sizes of Pipe

Temperature Difference Between Steam in Pipe and Air Surrounding Pipe

Pipe Size, Inches	50°	100°	150°	200°	250°	300°	350°	400°	450°	500°
	Per Cent Efficiencies									
½	76.8%	78.9%	80.6%	82.1%	83.4%	84.6%	85.8%	87. %	88.1%	89.1%
¾	79.7	81.7	83.1	84.5	85.6	86.6	87.6	88.6	89.6	90.5
1	81.9	83.5	84.8	86.1	87.1	88.	88.9	89.8	90.7	91.5
1¼	83.5	85.	86.2	87.3	88.2	89.1	89.9	90.8	91.6	92.3
1½	84.7	86.1	87.2	88.3	89.1	89.9	90.6	91.4	92.2	92.8
2	86.5	87.7	88.7	89.6	90.3	91.	91.7	92.4	93.1	93.7
2½	87.5	88.7	89.6	90.5	91.1	91.7	92.4	93.	93.5	94.2
3	88.4	89.5	90.3	91.2	91.7	92.3	92.9	93.5	94.1	94.6
3½	89.	90.	90.8	91.6	92.1	92.7	93.3	93.8	94.4	94.8
4	89.5	90.5	91.2	92.	92.5	93.	93.6	94.1	94.6	95.1
4½	89.9	90.8	91.5	92.2	92.8	93.3	93.8	94.3	94.8	95.2
5	90.2	91.1	91.8	92.5	93.	93.5	94.	94.5	95.	95.4
6	90.7	91.5	92.2	92.9	93.3	93.8	94.3	94.8	95.2	95.6
7	91.1	91.9	92.5	93.2	93.6	94.1	94.5	95.	95.4	95.8
8	91.3	92.1	92.7	93.4	93.8	94.2	94.7	95.1	95.5	95.9
9	91.5	92.3	92.9	93.5	93.9	94.4	94.8	95.2	95.6	96.
10	91.7	92.5	93.	93.6	94.	94.5	94.9	95.3	95.7	96.1

TABLE NO. 7.

LIST PRICES OF PIPE COVERING

SUBJECT TO DISCOUNT

INSIDE DIAMETER OF PIPE

	½"	¾"	1"	1¼"	1½"	2"	2½"	3"	3½"	4"	4½"	5"	6"	7"	8"	9"	10"	12"	*14"	16"	18"	20"	24"	30"
	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$
Standard Thick..	.22	.24	.27	.30	.33	.36	.40	.45	.50	.60	.65	.70	.80	1.00	1.10	1.20	1.30	1.85	2.10	2.35	2.60	2.85	3.30	4.00
1½" Thick..	.46	.49	.52	.56	.60	.64	.70	.76	.82	.88	.94	1.00	1.10	1.20	1.35	1.50	1.65	1.85	2.10	2.35	2.60	2.85	3.30	4.00
2" Thick	.75	.80	.85	.90	.95	1.00	1.05	1.15	1.25	1.35	1.45	1.55	1.70	1.85	2.00	2.20	2.40	2.70	3.00	3.30	3.60	4.00	4.50	5.50
**Double Standard Thick..	.65	.70	.75	.80	.85	.90	1.00	1.10	1.20	1.40	1.50	1.60	1.80	2.25	2.50	2.70	2.90	4.10	4.60	5.10	5.60	6.00	7.00	8.40
3" Thick Broken Joints..	1.20	1.35	1.40	1.45	1.55	1.65	1.75	1.90	2.05	2.20	2.35	2.50	2.70	2.90	3.15	3.40	3.65	4.10	4.60	5.10	5.60	6.00	7.00	8.40

	½"	¾"	1"	1¼"	1½"	2"	2½"	3"	3½"	4"	4½"	5"	6"	7"	8"	9"	10"
	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$	$
Elbows 90° & 45°..	.30	.30	.30	.30	.30	.36	.42	.48	.54	.60	.72	.90	1.30	1.80	2.40	3.00	3.60
Tees.............	.36	.36	.36	.36	.36	.42	.48	.54	.60	.75	.90	1.20	1.60	2.20	3.00	3.80	4.60
Crosses...........	.48	.48	.48	.48	.48	.54	.60	.70	.80	.95	1.10	1.50	2.00	2.80	3.60	4.40	5.20
Globe Valves......	.54	.54	.54	.54	.54	.60	.78	.96	1.20	1.50	1.85	2.25	2.80	3.60	4.40	5.30	6.20
Flange Covers.....	.50	.50	.50	.50	.50	.60	.70	.80	.90	1.00	1.30	1.60	1.90	2.20	2.50	2.90	3.30

These pipe insulations are supplied in sections three feet long, canvased and with brass lacquered bands.

*85% Magnesia is made in Standard (approx. 1"), 1½", 2", Double Standard Thick and 3" (broken joint) thicknesses. For pipe sizes from and including 14" in diameter it is furnished in segmental form.

Asbesto-Sponge Felted Insulation is made in thicknesses from ½" to 3"; thicknesses 1" and under use list prices for standard thick.

Asbestocel and Air-Cell Insulations are made in ½", ¾", 1", 1½", 2", and 3" thicknesses; for thicknesses 1" and under use list prices for standard thick.

Zero Insulation is made in one thickness only, approximately 1¼". Use standard thick list prices. Prices on Zero Insulation for fittings on request.

Anti-Sweat Insulation is made only in ½", ¾" and 1" thicknesses; use standard thick list prices for all thicknesses.

Fittings not made for 85% Magnesia or Wool Felt Insulations.

**Applies only to 85% Magnesia.

APPENDIX IV.

THE FLOW OF STEAM IN PIPES.

30 - THE BABCOCK FORMULA.

Reliable experimental data on the friction loss of steam under various pressures, and particularly when superheated, are unfortunately lacking.

The most generally accepted formula for steam flow is Babcock's:

$$p = 0.000,131 \left(1 + \frac{3.6}{d}\right) \times \frac{w^2 L}{D d^5} \qquad \text{formula (1)}$$

The use of this formula may be greatly facilitated by rearrangement to the forms:

$$p = w^2 L V F \qquad \text{formula (2)}$$

or

$$p = \frac{w^2 L F}{D} \qquad \text{formula (3)}$$

Nomenclature:

- p = pressure loss, lb. per sq. in.
- w = steam flow, lb. per min.
- L = length of pipe or section, feet.
- D = average specific density of steam, lb. per cu. ft.
- V = average specific volume of steam, cu. ft. per lb.

  Note: For superheated steam use D and V for the <u>superheated</u> steam - not the saturated value. (See calculation, paragraph 32).

- d = inside diameter of pipe, inches.
- F = a factor which is a function of the pipe diameter only, =

  $$0.000,131 \frac{d + 3.6}{d^6} \text{ (see Table No. 8).}$$

Since the value F is a function of only one variable, viz. pipe diameter, it may be tabulated. In Table No. 8 columns 1 and 4 indicate regularly manufactured nominal pipe sizes from 1/2 inch upward. Columns 2 and 5 show the "actual" inside diameters corresponding to columns 1 and 4. Columns 3 and 6 indicate the values of factor "F" to be used in formulae 2 and 3 corresponding to columns 1 and 4.

31 - STANDARD AND EXTRA STRONG PIPES.

It should be noted particularly that column 3 applies only to standard

TABLE NO. 8.

FACTOR "F" FOR THE MODIFIED BABCOCK FORMULA.

Standard Weight Pipe			Extra Heavy Pipe		
Nominal Size, Inches	Actual Inside Diam.	Pressure Loss Factor, F	Nominal Size, Inches	Actual Inside Diam	Pressure Loss Factor, F
1	2	3	4	5	6
½	0.622	9551. $\times 10^{6}$	½	0.546	20.51 $\times 10^{-3}$
¾	0.824	1817. "	¾	0.742	3408. 10^{6}
1	1.049	457.1 "	1	0.957	777.1 "
1¼	1.380	94.32 "	1¼	1.278	146.7 "
1½	1.610	39.14 "	1½	1.500	58.65 "
2	2.067	9.519 "	2	1.939	13.65 "
2½	2.469	3510. $\times 10^{9}$	2½	2.323	4938. $\times 10^{-9}$
3	3.068	1047. "	3	2.900	1432. "
3½	3.548	469.4 "	3½	3.364	629.5 "
4	4.026	231.6 "	4	3.826	310.1 "
4½	4.506	126.9 "	4½	4.290	165.8 "
5	5.017	68.51 "	5	4.813	88.66 "
6	6.065	25.44 "	6	5.761	33.54 "
7	7.023	11.60 "	7	6.625	15.84 "
8	8.071	5531 $\times 10^{-12}$	8	7.625	7482. $\times 10^{12}$
8	7.981	5870. "	9	8.625	3890. "
9	8.941	3216. "	10	9.750	2036. "
10	10.192	1612 "	11	10.750	1217. "
10	10.136	1659. "	12	11.750	761.1 "
10	10.020	1763 "	13	13.000	450.5 "
11	11.000	1080. "	14	14.000	306.2 "
12	12.090	658.2 "	15	15.000	213.9 "
12	12.000	684.4 "	17 OD	16.000	153.0 "
13	13.250	407.9 "	18 "	17.000	111.8 "
14	14.250	279.2 "	20 "	19.000	62.93 "
15	15.250	196.3 "	22 "	21.000	37.57 "
17 OD	16.214	143.0 "	24 "	23.000	23.51 "
18 "	17.182	105.8 "			
20 "	19.182	59.91 "			

weight pipe, and column 6 to extra strong pipe. It might be supposed from the nearness of agreement between the actual diameters of standard and extra strong pipes that no distinction between the two need be made when calculating pressure loss. That this supposition is erroneous may be seen by comparing columns 3 and 6 of the table. Since the pressure loss is directly proportional to factor "F", the relative friction losses in the two kinds of pipe for a given nominal size may be observed by the ratio of the two values of factor "F". Such a comparison reveals the fact that if column 3 be used for problems involving extra strong pipe, the error would be -10 percent for 12-inch pipe, -55 percent for 1/2-inch pipe, and the average error for all sizes from 1/2-inch to 12-inch inclusive would be -28 percent.

32 - SUPERHEATED STEAM.

Since experimental data on the flow of superheated steam are not available, it is necessary to deduce some relation from the flow of saturated steam.

There are two conditions which differ for wet and superheated steam, viz., surface condition and velocity. The difference in surface condition, is that in the case of superheated steam the inner pipe surface is dry, whereas in the case of wet steam the surface is supposedly flushed with a film of water. Several theories are based upon this fact. One is that the water on the wetted surface fills up the irregularities in the pipe surface, resulting in a lower coefficient of friction for wet steam than for dry or superheated steam. Another theory, exactly contradictory to the foregoing, is that the moisture presents a more or less viscous filament which imposes a drag on the flowing stream of vapor, and consequently the coefficient of friction should be greater for wet steam.

The correctness of either of these theories has not been proved, and it is possible that both of the effects, acting in conjunction, counterbalance one another, resulting in the same coefficient of friction for superheated or dry steam as for wet steam. The difference, in any event, is undoubtedly small, and since the effect of velocity is of considerable magnitude, the effect of surface condition may well be disregarded.

The effect of the velocity of flow upon the pressure loss is not a simple one. Friction for most fluids is supposed to vary as the square of the velocity, but a study of the Babcock formula shows that this relation is not universal for steam. Velocity is affected by: (1) the weight of steam flowing, (2) the cross-sectional area of the pipe, and (3) the specific volume of the steam.

(1) Inspection of the Babcock formula indicates that pressure loss varies as w^2 , and since the velocity is proportional to w, it follows that the pressure loss varies as the square of the velocity, which would be expected.

(2) By plotting factor "F" against velocity on logarithmic cross-section paper, it is found that the friction varies approximately as the 2.6 power of the velocity, or somewhat greater than the square. The probable explanation is that as pipe size is decreased, the mean hydraulic radius decreases, presenting a comparatively greater frictional surface.

(3) A set of calculations with saturated steam, making steam pressure the only variable, indicates that friction varies as the first power of the

velocity. The explanation of this unexpected fact probably lies in the fact that as the pressure is decreased, although the specific volume and hence velocity are greater, the density is correspondingly decreased, and the number of molecules of steam in contact with unit surface of pipe is proportionately less.

In view of the relation found in the preceding paragraph, it is the logical conclusion that, since the effect of surface condition may be disregarded, the friction for various amounts of superheat will vary as the first power of the velocity.

The solution for pressure loss with superheated steam may be made in either of two ways, as will be explained.

(1) Since the velocity of flow varies as the specific volume of the steam, either of the formulae, (1), (2), or (3) may be used directly by merely substituting the proper value of specific density or specific volume for "D" or "V" respectively from the superheated steam tables. This is the preferred method when superheated steam tables are at hand.

(2) A careful examination of the properties of superheated steam indicates that the increase in volume at any given pressure is very nearly 16 percent for every 100 degrees of superheat. This assumption is so nearly exact that the worst error due to its use is less than two percent, which is beyond criticism since there is four percent variance between the experimental coefficients of friction as determined by Babcock and Carpenter. The second method of handling superheated steam, then, is to use any reliable formula or chart designed for saturated steam, and increase the pressure loss 16 percent for every 100 degrees of superheat.

33 - EXAMPLE OF SOLUTION BY THE FORMULA.

Data:

Steam pressure - 225 lb. per sq. in. abs.
Superheat - 150 deg. F.
Steam Flow - 2000 lb. per min.
Pipe - 12-inch extra strong.

Determine: Pressure loss per 100 feet of pipe.
Solution: Use formula (2)

$w = 2000 \qquad w^2 = 4{,}000{,}000.$
$L = 100$ feet.
$V = 2.56$ cu. ft. per lb. (from steam tables)
$F = 764.1 \times 10^{-12}$ (from Table No. 8).
$p = 4{,}000{,}000 \times 100 \times 2.56 \times 764.1 \times 10^{-12} =$
0.782 lb. per sq. in.

34 - GRAPHIC CHART FOR THE BABCOCK FORMULA.

Although the solution of the modified formulae is quite simple when Table No. 8 and a slide rule are available, the logarithmic chart, Fig. 6, permits a quick and easy solution of flow problems, and it is preferred in most cases.

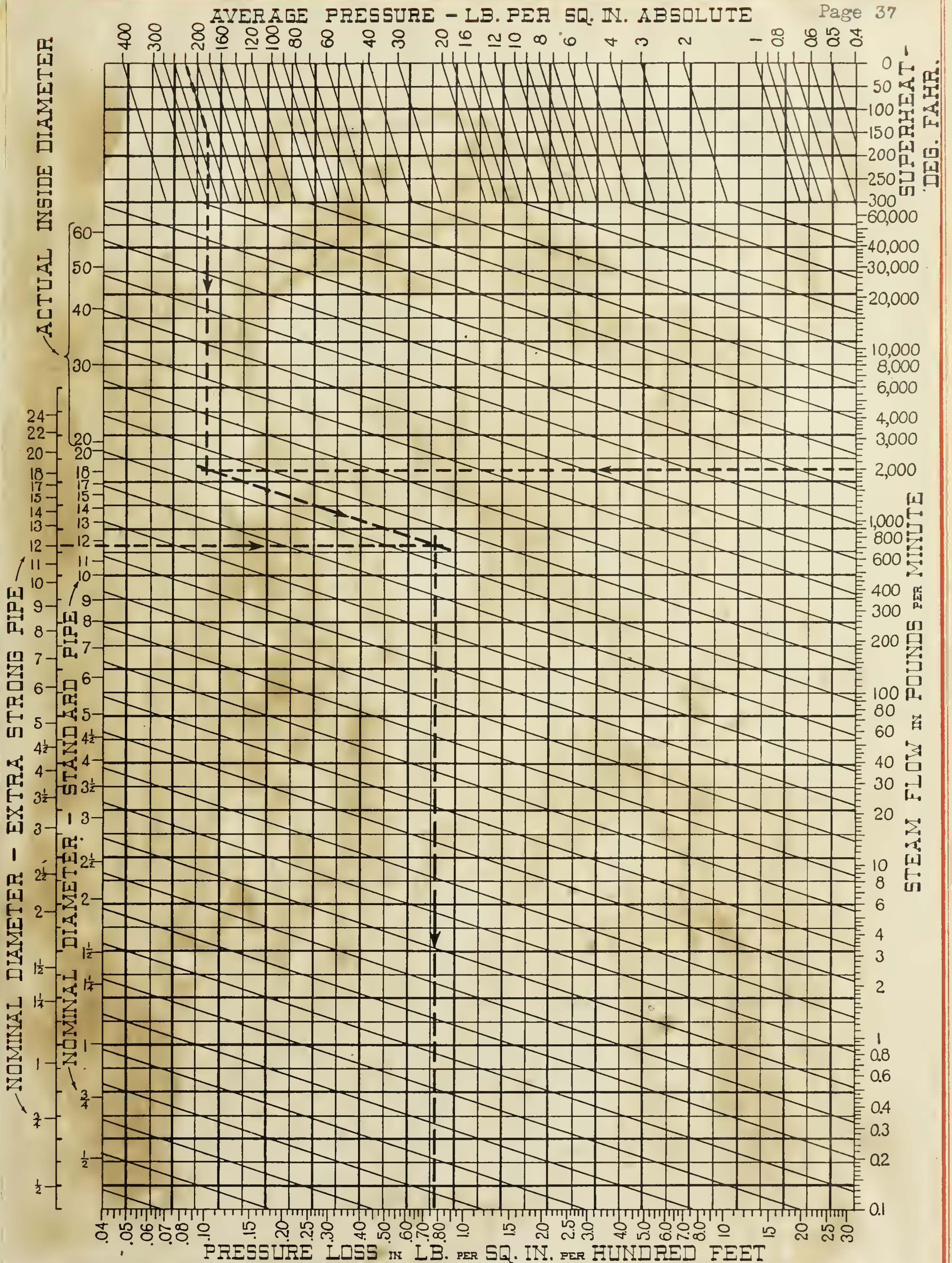


FIG. 6. - GRAPHIC STEAM FLOW CHART.

The error involved in its use is not more than three percent if reasonable care is exercised in the solution.

This form of chart was first published in 1912 by H. V. Carpenter. The original chart, however, was limited in its direct application to saturated steam and standard weight pipe, and the range of the quantity of flow scale was not extensive enough for many modern problems. Fig. 6 was drawn by the author to meet these criticisms and the chart as presented is practically universal in its application.

35 - EXAMPLE OF GRAPHICAL SOLUTION.

The heavy dash lines on the chart indicate the solution of the same problem as solved under "Example of Solution by the Formula". The solution is self-explanatory and needs no further comment.